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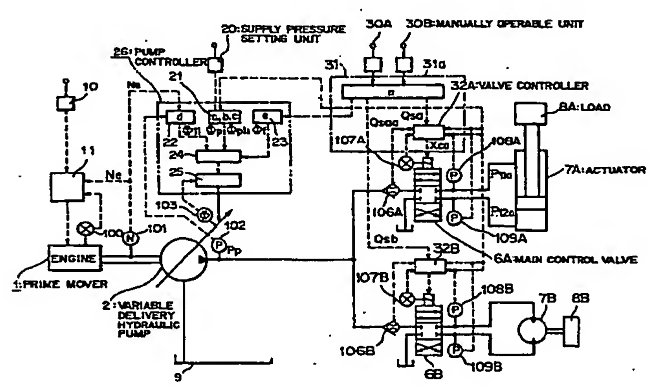
(54) CONTROL SYSTEM FOR CONSTRUCTION MACHINE

(57) The present invention relates to a control apparatus for a construction machine. When operation commands from manually operable means (30A, 30B) and results of detection from detection means (102) are received, requested flow rate information (Qra, Qrb) to actuators (7A, 7B) set by the manually operable means (30A, 30B) and working fluid supply flow rate information from working fluid supply means are compared with each other and optimal supply flow rates to the actuators (7A, 7B) are determined based on results of the comparison by a distributor function of valve control means (31) provided in control means, and valve means is controlled with the optimal supply flow rates.

FIG. 6



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Description

FIELD OF THE INVENTION

This invention relates to a control apparatus for a construction machine suitable for use with a hydraulic excavation machine, a hydraulic shovel or a like machine.

BACKGROUND OF THE INVENTION

Conventionally, in a fluid pressure driving system for a construction machine such as a hydraulic shovel, the opening of a main control valve is remotely controlled using a hydraulic or electromagnetic hydraulic pilot valve to adjust the flow rate of working fluid to actuators (for example, in a hydraulic shovel, to a boom cylinder, a stick cylinder and so forth).

However, in order to allow an operator to drive a plurality of actuators, to which different loads are applied, simultaneously as in a simultaneous operation in accordance with an intention of the operator, both of a manual operation of a control lever and adjustment of a delivery pressure and a delivery flow rate of an engine driven variable delivery fluid pressure pump serving as a fluid pressure energy supply source must be performed, and very high skill is required.

Thus, a load sensing type 1-pump system wherein a simultaneous operation can be performed comparatively readily, that is, a liquid pressure driven system which adopts main control valves (the difference in pressure across each of the valves is constant and the flow rate increases in proportion to the opening of the valve) of the closed center type connected in parallel to each other, has been proposed recently.

Here, a representative example of such a load sensing type 1-pump system as mentioned above will be described with reference to FIG. 7. FIG. 7 is a schematic view showing a construction of a hydraulic driving apparatus disclosed in International Publication No. WO93-16285 (Japanese Patent Application No. Heisei 5-510414).

Operation amount detectors 450A and 450B set electric signals in response to operation amounts of manually operable levers 405A and 405B and output the electric signals to valve flow rate control apparatus 411A and 411B, respectively.

Meanwhile, flow rates of working fluid supplied from a variable delivery hydraulic pump 401 to a plurality of hydraulic actuators 403A and 403B via pressure compensated flow control valves 440A and 440B are detected by flow rate detectors 410A and 410B, and this detection information is fed back to the valve flow rate control apparatus 411A and 411B, respectively.

Then, control signals outputted from the valve flow rate control apparatus 411A and 411B to a pump tilting control apparatus 412 are used to control a pump regulator 420 for operating a volume variation mechanism 401a of the variable delivery hydraulic pump 401 and effect direction control and flow rate control of the pressure compensated flow control valves 440A and 440B by means of the control apparatus 411A and 411B, respectively.

In particular, the prior art apparatus is constructed as a system of a flow rate servo type and as a hydraulic source system of the energy saving type (which delivers a pump flow rate lower than a requested flow rate) wherein the opening of the flow rate control valve of the highest load pressure is made maximum to minimize the pressure loss of the flow rate control valve.

Meanwhile, FIG. 8 is a schematic view showing a construction of a hydraulically driven apparatus disclosed in International Publication No. WO93-18308 (Japanese Patent Laid-Open Application No. Heisel 5-514375). The hydraulically driven apparatus effects direction control and flow rate control of working oil supplied from a variable delivery hydraulic pump 501 to a plurality of actuators 502a and 502b using flow rate control valves 503a and 503b, respectively.

It is to be noted that the flow rate control valves 503a and 503b include spools which are displaced in response to values of currents transmitted thereto from a controller 510 via solenoid wiring lines 511, 512, 513 and 514.

Meanwhile, an unload valve 507 is connected to the hydraulic pump 501, and when a pressure difference between the delivery pressure of the variable delivery hydraulic pump 501 and a maximum load pressure extracted via a shuttle valve 506 exceeds a predetermined value, the unload valve 507 is opened so that working oil delivered from the hydraulic pump 501 is returned to a tank. It is to be noted that a difference pressure setting screw 507a is provided on the load pressure acting side of the unload valve 507.

A fixed orifice 508 for generating a control pressure in response to a flow rate of the working fluid flowing out from the unload valve 507 is connected to the downstream of the unload valve 507, and the control pressure generated by the fixed orifice 508 is detected by a pressure sensor 515.

Further, a control apparatus for the variable delivery hydraulic pump 501 is composed of a pump regulator 509, the controller 510, the pressure sensor 515, a displacement sensor 516 and so forth and is constructed such that, when the control pressure generated by the fixed orifice 508 becomes high, the delivery flow rate of the hydraulic pump 501 is decreased, but when the control pressure becomes low, the delivery flow rate is increased.

Furthermore, a directional control valve 530 is connected in parallel to the unload valve 507 at a position on the upstream with respect to the fixed orifice 508. A solenoid operated proportional pressure reducing valve 531 is control-

led by a signal outputted from the controller 510 in response to an operation signal from a manually operable lever apparatus 505 to control the pilot hydraulic pressure from a pilot hydraulic pressure source 521 to the directional control valve 530.

Consequently, the directional control valve 530 is controlled such that, when the operation amount of a manually operable lever 504 is small, the opening area of the directional control valve 530 is large, and as the operation amount of the manually operable lever 504 increases, the opening area decreases.

Accordingly, the load sensing control by the unload valve 507 and the bleed-off control by the directional control valve 530 are selectively performed in response to the operation amount of the manually operable lever apparatus 505, and the plurality of actuators 502a and 502b are driven by the flow rate control which makes most of characteristics of the two controls.

Further, a hydraulically driven control apparatus which includes a load sensing system which in turn includes meter-in and meter-out separation valves and a pressure compensation valve for setting pressure differences across the valves.

However, the system concepts of such conventional liquid pressure driven systems as described above are all directed to saving of energy, and components of a conventional hydraulic apparatus body and a conventional hydraulic apparatus adjustment system are individually collected to construct a system. In particular, in the control of actuators, stress is placed on pump control of a high transmission efficiency, and for control valves (for example, the pressure compensated flow control valves 440A and 440B or the flow rate control valves 503a and 503b described hereinabove), stress is placed on the directional changing over function because they exhibit a comparatively low pressure loss.

Accordingly, mutual interference between the liquid pressure source and the valve flow rate adjustment system (mutual interference signifies that, in a simultaneous operation, the flow rate of an actuator is varied because the pressure is varied by a variation of the load to another actuator) has not been augmented as yet, and the operability (particularly the fine operability) is insufficient.

Further, a main control valve for a flow rate adjustment system is used only for flow rate adjustment, and it has not been considered to effect pressure control by feedback using only a control valve.

However, in such a fluid pressure driven system, a high inertial load acts frequently, and in such an instance, the fluid pressure driven system has a resonance frequency based on a piping characteristic and the inertial load (which varies in response to the load or the posture of the machine) and has a problem in that the system is much likely to be vibrated.

Further, by such a sudden load variation as occurs when a working apparatus (bucket or the like) violently collides with a hard substance such as a rock or a sudden manual operation when an emergent situation occurs, lower to higher harmonics are sometimes produced on a machine body. Consequently, the driving feeling is not good and improvement in workability cannot be anticipated.

The present invention has been made in view of such circumstances as described above, and it is an object of the invention to achieve improvement in manual operability, augmentation in driving feeling and improvement in workability of a construction machine.

DISCLOSURE OF THE INVENTION

In order to attain the object, a control apparatus for a construction machine of the present invention is characterized in that it comprises manually operable means manually operable by an operator, working fluid supply means including a hydraulic pump driven by a prime mover, driving means including a plurality of actuators driven by working fluid from the working fluid supply means, valve means including a plurality of control valves interposed between the driving means and the working fluid supply means for controlling the driving means, detection means including working fluid supply flow rate detection means for detecting a supply flow rate of the working fluid from the working fluid supply means, and valve control means for receiving an operation command from the manually operable means and a result of detection from the detection means and controlling the valve means by a distributor function of comparing requested flow rate information to the actuators set by the manually operable means with the working fluid supply flow rate information from the working fluid supply means and determining optimal supply flow rates to the actuators in response to a result of the comparison.

With the construction, operation signals from the valve control means having the distributor function are outputted as supply flow rate setting commands to the plurality of control valves, and the actuators operate with a hydraulic pressure from the hydraulic pump. The valve control means compares, by the distributor means thereof, requested flow rate information to the actuators set by the manually operable means with working fluid supply flow rate information from the working fluid supply means and determines optimal supply flow rates to the actuators in response to a result of the comparison. Consequently, distribution of the requested flow rates to the actuators can be realized accurately.

The valve control means may include a distributor which outputs the requested flow rate signals to the actuators by the manually operable means as actuator flow rate setting signals when a sum total of the requested flow rates is lower than the working fluid supply flow rate, but outputs values obtained by multiplying the requested flow rates to the actu-

ators by a coefficient smaller than 1 as actuator flow rate setting signals when a sum total of the requested flow rates is higher than the working fluid supply flow rate.

Thus, by the distributor, based on operation signals outputted from the operation system to the main control valves, an actuator flow rate distribution required by an operator can be realized with a delivery flow rate of the pump irrespective of loads to the actuators. Consequently, improvement in operability, improvement particularly in simultaneous operability and fine operability, can be anticipated, and improvement in workability can be anticipated and the skill of the operator can be exhibited sufficiently.

The coefficient smaller than 1 may have information obtained by normalization of the working fluid supply flow rate with the sum total of the requested flow rate information.

The actuator flow rate setting signals set by the distributor may be set for each work mode of the construction machine.

In this instance, optimal supply flow rates to the actuators are determined in response to a work mode, and distribution of the requested flow rates to the actuators can be realized accurately. Consequently, the plurality of actuators can be driven at the same time in accordance with a will of the operator without requiring a high skill, and the efficiency in working is improved.

The control apparatus for a construction machine may be constructed such that the detection means includes manipulation detection means for detecting operation conditions of the valve means, and the valve control means includes correction means for receiving results of the detection from the manipulation detection means and correcting the distributor function.

Further, the manipulation detection means may include spool position sensors for measuring and feeding back spool positions of the control valves, load sensing load pressure sensors for measuring and feeding back load pressures, and flow rate sensors for measuring and feeding back flow rates supplied to the actuators. By such construction, the spool positions of the control valves can be controlled with a high degree of accuracy.

Each of the load sensing load pressure sensors may include a band-pass filter at an output portion thereof, and this can prevent an overshoot in the spool position control.

The working fluid supply means may include an accumulator for accumulating the working fluid on a delivery side of the hydraulic pump. Further, the working fluid supply means may include an unload valve for bypassing a delivery flow rate of the hydraulic pump in a no-load condition when a capacity of the accumulator exceeds a predetermined amount.

By the construction described above, a supply pressure variation can be suppressed low against a variation in manual operation, a large variation in flow rate or a sudden variation in flow rate, and mutual interference in pressure variation between the actuators is eliminated and lower harmonics or fluctuations of a construction machine structure can be suppressed and besides improvement in operability and augmentation in driving feeling of an operator in a cab can be anticipated. Further, an unnecessary pump flow rate is allowed to bypass by the unload valve, and saving of the fuel cost can be achieved. Furthermore, a flow rate higher than the pump delivery flow rate can be supplied temporarily by the fluid pressure accumulated in the accumulator, and this allows improvement in productivity.

The control apparatus for a construction machine may be constructed such that the unload valve is provided in parallel to a working fluid supply path on the delivery side of the hydraulic pump while the accumulator is provided in parallel at a portion of the working fluid supply path on a downstream side with respect to a connection point of the unload valve to the working fluid supply path, and a check valve for preventing a back flow from the accumulator is interposed in a portion of the working fluid supply path between the connection portion of the unload valve and a connection portion of the accumulator to the working fluid supply path.

The manually operable means may include a supply pressure setting unit for keeping a pump delivery pressure of the hydraulic pump fixed. This allows so-called fixed supply pressure operation wherein a pump delivery pressure command signal programmed in advance is provided in response to contents of a work, and can improve the workability and allows an operation which can esteem the skill of the operability.

The control apparatus for a construction machine may be constructed such that the working fluid supply means includes an accumulator for accumulating the working fluid on a delivery side of the hydraulic pump, and the valve control means includes a distributor which outputs the requested flow rate signals to the actuators by the manually operable means as actuator flow rate setting signals when a sum total of the requested flow rates is lower than the working fluid supply flow rate, but outputs values obtained by multiplying the requested flow rates to the actuators by a first coefficient smaller than 1 as actuator flow rate setting signals when the sum total of the requested flow rates is higher than the working fluid supply flow rate, and calculates a total of an accumulation supply flow rate of the accumulator and the working fluid supply flow rate as an allowable supply flow rate and outputs values obtained by multiplying the requested flow rates to the actuators by a second coefficient having information obtained by normalization of the allowable supply flow rate with the sum total of the requested flow rates as actuator flow rate setting signals.

The control apparatus for a construction machine may be constructed such that the first coefficient has information obtained by normalization of the working fluid supply flow rate with the sum total of the requested flow rates, and such that at least one of the first coefficient and the second coefficient is set for each work mode of the construction machine.

The control apparatus for a construction machine may be constructed such that the detection means includes power supply side detection means for detecting an operation condition of the working fluid supply means, and the control means includes power supply side control means for receiving a result of the detection from the power supply side detection means and controlling the working fluid supply means.

Further, power supply side detection means may include a rotation condition sensor for detecting a rotation condition of the prime mover, an output power sensor for detecting an output power condition of the prime mover, and a working fluid pressure sensor for detecting a pressure of the working fluid from the working fluid supply means.

Meanwhile, another control apparatus for a construction machine of the present invention is characterized in that it comprises manually operable means manually operable by an operator, at least one variable delivery liquid pressure pump driven by an engine, a plurality of liquid pressure actuators driven by pressure fluid delivered from the variable delivery liquid pressure pump, a plurality of main control valves interposed between the liquid pressure actuators and the variable delivery liquid pressure pump for controlling flow rates and directions to the liquid pressure actuators, an accumulator provided in a liquid path between the variable delivery liquid pressure pump and the main control valves for accumulating the pressure fluid, an unload valve provided in the liquid path between the variable delivery liquid pressure pump and the main control valves for allowing bypassing of a delivery flow rate of the variable delivery liquid pressure pump in a no-load condition when a capacity of the accumulator approaches a maximum value of the capacity, a distributor including first calculation means for outputting requested flow rate signals to the actuators by the manually operable means as they are as actuator flow rate setting signals when a sum total of the requested flow rates to the actuators by the manually operable means is lower than a delivery flow rate of the variable delivery liquid pressure pump, but outputting, when the sum total of the requested flow rates is higher than the pump delivery flow rate, values obtained by multiplying the requested flow rates to the liquid pressure actuators by a value obtained by dividing the pump delivery flow rate by the sum total of the requested flow rates as actuator flow rate setting signals and second calculation means for multiplying the requested flow rates to the actuators by a value obtained by dividing an allowable supply flow rate calculated as a total of an accumulation supply flow rate of the accumulator and the pump delivery flow rate by the sum total of the requested flow rates and outputting results of the multiplication as actuator flow rate setting signals, a supply pressure setting unit provided for the manually operable means for keeping the pump delivery output fixed, a valve controller for receiving the actuator flow rate setting signals from the distributor and supplying operation signals to the main control valves, a manipulation side sensor group provided for the valve controller and including spool position sensors for measuring and feeding back spool positions of the main control valves, load sensing load pressure sensors with a band-pass filter for measuring and feeding back load pressures, and flow rate sensors for measuring and feeding back flow rates supplied to the liquid pressure actuators, a power supply side sensor group including a rotation speed sensor for measuring an engine speed, a rack opening sensor for measuring a rack opening of an engine fuel pump, a tilting angle sensor for measuring a pump tilting angle, a delivery pressure sensor for measuring a pump delivery pressure, a supply pressure sensor for measuring a system supply pressure and an accumulator capacity sensor for measuring a capacity of the accumulator, first command means for generating a tilting angle command signal for the variable delivery liquid pressure pump based on a sum of a difference between a pressure set by the supply pressure setting unit and the feedback signal from the supply pressure sensor and an integrated value of the difference, second command means for selecting a maximum signal from among the supply pressure setting unit and the load sensing load pressure sensors, determining a value obtained by addition of a fixed value to a value of the maximum signal as a command signal when the value of the maximum signal continues for more than a fixed period of time and generating a tilting angle command signal for the variable delivery liquid pressure pump based on a sum of a difference between the command signal and the feedback signal from the supply pressure sensor and an integrated value of the difference, third command means for generating a signal to open the unload valve to allow bypassing of the delivery flow rate of the variable delivery liquid pressure pump in a no-load condition when a supply pressure rises higher by a certain value than a preset value and the capacity of the accumulator is in the proximity of a maximum value thereof, but to close the unload valve when the supply pressure drops lower by a certain value than the preset value or the capacity of the accumulator drops to a value in the proximity of a minimum value thereof, fourth command means for generating an allowable tilting angle command signal for the variable delivery liquid pressure pump within a range of an output power of the engine as a function of the output delivery power of the engine and an efficiency characteristic of the engine-pump, fifth command means for generating a tilting angle command signal for the variable delivery liquid pressure pump for securing a pump flow rate which increases in proportion to a flow rate request of an operator, and a pump controller for selecting a lowest one of the generated command signals as a tilting angle command signal for the variable delivery liquid pressure pump and positioning a pump tilting angle based on a difference between the selected tilting angle command signal and the feedback signal from the tilting angle sensor.

With the construction described above, improvement in response, safety and flow rate control accuracy of the system can be anticipated. By such addition of the high speed pressure controlling function of the main control valves, higher harmonics of a front working machine or a machine body can be suppressed, and the simultaneous operability, the fine operability and the driving feeling of an operator in a cab can be augmented.

Further, by liquid pressure electronic control systematization wherein function allotments between liquid pressure

apparatus and electronically controlled apparatus are made definite compositely improving conventional liquid pressure driven systems from the systematic point of view, improvement in operability, augmentation in driving feeling and improvement in workability can be achieved.

A further control apparatus for a construction machine of the present invention is characterized in that it comprises manually operable means manually operable by an operator, a hydraulic pump driven by a prime mover, a plurality of actuators driven by working fluid from the hydraulic pump, a plurality of control valves for controlling the actuators, and valve control means for comparing requested flow rate information to the actuators set by the manually operable means with working fluid supply flow rate information from the hydraulic pump, determining optimal supply flow rates to the actuators based on results of the comparison and controlling the valve means with the optimal supply flow rates.

By the construction, the plurality of actuators can be driven at the same time in accordance with a will of an operator without requiring a high skill, and the efficiency in working is improved.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a schematic hydraulic circuit diagram showing essential part of a control apparatus for a construction machine as a first embodiment of the present invention;
FIG. 2 is a schematic block diagram showing a general construction of the control apparatus for a construction machine as the first embodiment of the present invention;
FIG. 3 is a block diagram showing a control system for a manipulation system in the control apparatus for a construction machine as the first embodiment of the present invention;
FIG. 4 is a view showing a modification to the control apparatus for a construction as the first embodiment of the present invention and is a view showing a data table of coefficients set for each work mode;
FIG. 5 is a view showing another example of a hydraulic circuit applicable to the control apparatus for a construction machine as the first embodiment of the present invention;
FIG. 6 is a schematic hydraulic circuit diagram of a control apparatus for a construction machine as a second embodiment of the present invention;
FIG. 7 is a hydraulic circuit diagram showing an example of a conventional liquid pressure driven circuit for a construction machine; and
FIG. 8 is a hydraulic circuit diagram showing another example of a conventional liquid pressure driven circuit for a construction machine.

BEST FORMS IN EMBODYING THE INVENTION

In the following, the present invention will be described in connection with embodiments thereof with reference to FIGS. 1 to 6.

(1) Description of the First Embodiment

Referring to FIG. 1, an apparatus shown includes a diesel engine (hereinafter referred to simply as engine) 1 as a prime mover, a variable delivery hydraulic pump (hereinafter referred to simply as hydraulic pump) 2 serving as a fluid pressure pump driven by the engine 1, and a plurality of hydraulic actuators 7A and 7B driven by high pressure working fluid delivered from the hydraulic pump 2.

A plurality of main control valves (closed center valves) 6A and 6B are interposed between the hydraulic pump 2 and the hydraulic actuators 7A and 7B so that the directions and the flow rates of working oil to be supplied to the hydraulic actuators 7A and 7B may be controlled in response to an operation command signal from a manual operation system operated by an operator.

In particular, the actuators 7A and 7B operate in response to manually operated conditions of manually operable levers 30A and 30B which serve as manually operable means.

Flow rate sensors 106A and 106B with a check valve are provided on the upstream sides of the main control valves 6A and 6B, respectively.

Further, connected in parallel in oil paths between the hydraulic pump 2 and the main control valves 6A and 6B are an unload valve 3 which allows bypassing of working fluid delivered from the hydraulic pump 2 to a hydraulic tank 9 when no load is applied and an accumulator 5 for accumulating working fluid delivered from the hydraulic pump 2 therein.

Here, a working fluid supply path (oil path) on the delivery side of the hydraulic pump 2 is, branched into two directions on the downstream side, and the unload valve 3 is provided for one of the two oil paths while the accumulator 5 is provided for the other oil path via a check valve 4. It is to be noted that the check valve 4 is provided to prevent a back flow of working fluid from the accumulator 5.

Further, the present apparatus includes control means for controlling operation of the actuators 7A and 7B, hydrau-

lic pump 2, main control valves 6A and 6B and so forth. Operation of the main control valves 6A and 6B from among those elements is controlled by valve control means 31 provided in the control means.

The valve control means 31 includes a distributor 31a which receives operation commands from the manually operable levers 30A and 30B and results of detection of sensors which will be hereinafter described, compares requested flow rate information to the actuators 7A and 7B set by the manually operable levers 30A and 30B with working fluid supply flow rate information of the hydraulic pump 2 and determines optimal supply flow rates to the actuators 7A and 7B in response to results of the comparison.

Then, the distributor 31a outputs, when the sum total of the requested flow rates for working fluid to the actuators 7A and 7B by the operation conditions of the manually operable levers 30A and 30B is lower than the delivery flow rate of the hydraulic pump 2, the requested flow rate signals to the actuators 7A and 7B by the manually operable levers 30A and 30B as they are as actuator flow rate setting signals. On the other hand, when the sum total of the requested flow rates is higher than the pump delivery flow rate, the distributor 31a multiplies the requested flow rates to the actuators 7A and 7B by a value α ($\alpha < 1$: first coefficient) obtained by dividing the pump delivery flow rate by the sum total of the requested flow rates, sets values obtained by the multiplication as working fluid requested amounts newly, and outputs the thus set requested flow rate signals as actuator flow rate setting signals.

Further, the distributor 31a calculates a sum of the accumulation supply flow rate of the accumulator 5 and the delivery flow rate from the hydraulic pump 2 as an allowable supply flow rate, multiplies the requested flow rates to the actuators 7A and 7B by a value β obtained by dividing the allowable supply flow rate by the sum total of the requested flow rates, sets values obtained by the multiplication newly as working fluid requested amounts, and outputs the requested flow rate signals as actuator flow rate setting signals.

The present apparatus further includes a supply pressure setting unit 20 for keeping the delivery pressure of the hydraulic pump 2 to a fixed level.

Further, spool position sensors 107A and 107B for detecting spool positions (that is, valve openings) are provided for the main control valves 6A and 6B, respectively, and feedback systems for spool positions, load pressure feedback systems with a band-pass filter and flow rate feedback systems are provided for valve controllers (correction means) 32A and 32B for outputting operation signals to the main control valves 6A and 6B, respectively.

In particular, manipulation detection means (or a manipulation side sensor group) including the flow rate sensors 106A and 106B with a check valve (which may be replaced by actuator velocity sensors or position sensors) for measuring and feeding back flow rates supplied to the actuators 7A and 7B, respectively, the spool position sensors 107A and 107B for measuring and feeding back spool positions (valve openings) of the main control valves 6A and 6B, respectively, and A port load pressure sensors 108A and 108B and B port load pressure sensors 109A and 109B for load sensing each including a band-pass filter 200 (refer to FIG. 3) for measuring and feeding back a load pressure on the output side of a main control valve is provided.

Further, power supply side detection means (or a power supply side sensor group) including a rack opening sensor (output sensor) 100 for measuring a rack opening of a fuel pump of the engine 1, an engine speed sensor (rotating condition sensor) 101 for measuring the speed of the engine 1, a pump delivery pressure sensor (working fluid pressure sensor) 102 for measuring the delivery output of the hydraulic pump 2, a pump tilting angle sensor 103 for measuring the tilting angle of the hydraulic pump 2, a supply pressure sensor 104 for measuring the supply pressure from the check valve 4 and an accumulator capacity sensor 105 for measuring the accumulator capacity is provided.

Further, the control means of the present apparatus includes power supply side control means (pump controller) 26. The pump controller 26 includes first command means a, second command means b, third command means c, fourth command means d and fifth command means e.

The pump controller 26 has a function of selecting and using a lowest one of signals emitted from the command means a, b, d and e mentioned above as a tilting angle command signal for the hydraulic pump 2 and positioning the pump tilting angle based on a difference between the selected tilting angle command signal and a feedback signal from the tilting angle sensor 103 of the hydraulic pump 2.

Here, the command means a to e mentioned above will be described. The first command means a is means for generating a tilting angle command signal Φp for the hydraulic pump 2 based on a sum of a difference between a pressure set by the supply pressure setting unit 20 and a feedback signal from the supply pressure sensor 104 and an integrated value of the difference, and has a function as a PI controller.

The second command means b is means for selecting a signal P1max having highest load information from within load information detected by the A port load pressure sensors 108A and 108B and the B port load pressure sensors 109A and 109B for load sensing of the main control valves 6A and 6B in addition to the supply pressure setting unit 20, determining, when this value continues for more than a fixed period of time, a value obtained by adding, to this value, a fixed value P10 as a command signal and generating a tilting angle command signal Φpls for the hydraulic pump 2 based on a sum of a difference between the command signal and a feedback signal from the supply pressure sensor 104 and an integrated value of the difference. Accordingly, also this second command means b has a function as a PI controller.

The third command means c is means for generating a signal for opening, when the supply pressure rises higher

than a certain value higher than a preset value and the accumulator capacity is in the proximity of the maximum value thereof, the unload valve 3 to allow bypassing of the delivery flow of the hydraulic pump 2 in a no-load condition, but closing the unload valve 3 when the supply pressure drops lower than a certain value lower than the preset value or the accumulator capacity drops to a value in the proximity of the minimum value thereof.

The fourth command means d is means for generating an allowable tilting angle command signal Φ11 for the hydraulic pump 2 within a range of the output power of the engine 1 as a function of three parameters including the output power of the engine 1, the delivery pressure of the hydraulic pump 2 and the efficiency characteristic of the engine-pump.

The fifth command means e is means for generating a tilting angle command signal Φf for the hydraulic pump 2 for securing a pump flow rate which increases in proportion to a flow rate request of an operator.

By the way, when notice is taken of functions of the present apparatus, it can be roughly divided into, as shown in FIG. 2, an operation system which is operated by an operator of the construction machine, a power supply system for supplying a hydraulic pressure, and a manipulation system for controlling the hydraulic pressure. Those systems will be hereinafter described with reference to FIGS. 1 and 2.

(a) Power Supply System

The diesel engine 1 which is a power source of the power supply system has an engine speed set corresponding to loads 8A and 8B by an engine throttle 10. In particular, an engine speed controller 11 outputs a command signal in response to an opening of the engine throttle 10, and positions the rack opening of the fuel pump in response to a feedback signal from the engine speed sensor 101 and another feedback signal from the rack opening sensor 100 of the fuel pump to automatically set the engine speed.

The power supply side control means (pump controller) 26 is formed from a supply pressure controller 21, an engine load limiter 22, a pump flow controller 23, a minimum signal selector 24 and a pump tilting angle regulator 25.

Then, the hydraulic pump 2 is controlled by the pump controller 26 so that it supplies a supply pressure conforming to the loads 8A and 8B similarly to the engine 1. A supply pressure signal is set by the supply pressure setting unit 20 and outputted to the supply pressure controller 21 for the hydraulic pump 2 and the accumulator 5.

In particular, the supply pressure controller 21 sets a tilting angle command signal Φp for the hydraulic pump 2 (first command means a) using a sum of a difference between a pressure set by the supply pressure setting unit 20 and a feedback signal from the supply pressure sensor 104 and an integrated value of the difference (PI control).

Meanwhile, the supply pressure controller 21 selects a maximum signal P1max from among the load pressure sensors 108A, 108B, 109A and 109B for load sensing, sets, when this value continues for more than a fixed period of time, a value obtained by adding to this value a fixed value P10, and sets a tilting angle command signal Φpls for the hydraulic pump 2 (second command means b) using a sum of a difference between the pressure set by the supply pressure setting unit 20 and a feedback signal from the supply pressure sensor 104 and an integrated value of the difference (PI control).

Further, the supply pressure controller 21 has, in addition to the tilting angle operation algorithm for the hydraulic pump 2 described above, an unload valve operation algorithm for opening, when the supply pressure rises higher than a certain value higher than a preset value and the accumulator capacity is in the proximity of the maximum value thereof, the unload valve 3 to allow bypassing of the variable delivery type pump flow in a no-load condition, but closing the unload valve 3 when the supply pressure drops lower than a certain value lower than the preset value or the accumulator capacity drops to a value in the proximity of the minimum value thereof (third command means c).

It is to be noted that the check valve 4 is provided to prevent high pressure working oil from flowing back from the accumulator 5 when the hydraulic pump 2 is put into an unloaded condition.

The engine load limiter 22 is provided in place of a conventional power mode selector and sets an allowable tilting angle command signal Φ11 for the hydraulic pump 2 within a range of the engine output power as a function of the pump capacity, an output Ne of the engine speed sensor 101, an output Pp of the pump delivery pressure sensor 102 and the efficiency characteristic of the engine-pump (fourth command means d).

The pump flow controller 23 is similar to a conventional positive flow rate control and outputs a tilting angle command signal Φf for the hydraulic pump 2 in order to secure a pump flow rate which increases in proportion to a flow rate request of an operator. The pump controller 26 can regard the tilting angle command signal Φf as one of feedforward signals (feedforward) (fifth command means e).

Then, the minimum signal selector 24 selects one of the pump tilting angle command signals Φp, Φpls, Φ11 and Φf generated from the means described above which sets the pump tilting angle to the lowest value.

The pump tilting angle regulator 25 receives an output signal of the minimum signal selector 24 as an input signal thereto and positions the tilting angle of the hydraulic pump 2 in response to a feedback signal from the pump tilting angle sensor 103.

As described above, the present power supply system is constructed as a power supply system which exhibits a large energy storage for securing a supply power to the manipulation system which will be hereinafter described and

hence has a so-called low-pass system characteristic.

(b) Manipulation System

The distributor 31a which functions as valve control means outputs actuator flow rate setting signals Qsa, Qsb, ... to the valve controllers 32A, 32B, ... in response to a situation of the power supply system when actuator flow rate request signals Qra, Qrb, ... are inputted from the manually operable levers (manually operable means) 30A and 30B (only two are shown here).

It is to be noted that the actuator flow rate request signals Qra, Qrb, ... are signals set independently of each other, and priority degrees of working oil to be supplied to the actuators 7A and 7B are set depending upon the magnitudes of requested flow rates represented by the signals.

Then, means for setting such actuator flow rate setting signals Qsa, Qsb, ... as mentioned above is available for each of the two following cases.

① When the sum total of the requested flow rates to the hydraulic actuators 7A, 7B, ... by the manually operable levers 30A, 30B, ... is lower than the delivery flow rate of the hydraulic pump 2, the requested flow rate signals to the actuators 7A and 7B by the manually operable levers 30A, 30B, ... are used as they are as actuator flow rate setting signals. In other words, Qsa = Qra , Qsb = Qrb , ...

On the other hand, when the sum total of the requested flow rates is higher than the pump delivery flow rate, the requested flow rates set by the manually operable levers 30A and 30B are multiplied by [pump delivery flow rate]/[sum total of requested flow rates] = α ($\alpha < 1$: first coefficient), and values obtained by the calculation are outputted as actuator flow rate setting signals to the valve controllers 32A, 32B, ... In other words, Qsa = αQra , Qsb = αQrb , ...

② An allowable supply flow rate Qs of the power supply system = function F(Xa, Qp, Ps) > Qp is calculated based on an output signal Xa of the capacity detection sensor 105 for the accumulator 5, a pump delivery flow rate signal Qp = function F(Ne, Φ, Pp) and an output signal Ps of the supply pressure sensor 104, and the requested signals to the actuator 7A, 7B, ... are multiplied by a value β ($\beta < 1$: second coefficient) obtained by dividing the allowable supply flow rate Qs by the sum total of the requested flow rates to make actuator flow rate setting signals. In other words, Qsa = βQra , Qsb = βQrb , ...

Consequently, an actuator flow rate distribution (including priorities) requested by an operator can be realized accurately, and the operability is improved very much and improvement in workability is anticipated.

The present manipulation system is characterized in that, different from a conventional hydraulically driven apparatus for a construction machine, the main control valves 6A and 6B have a higher response and multiple functions so that operation of the load driving hydraulic actuators 7A and 7B is controlled by electronically controlling a single main control valve to control all of the flow rate and pressure variations arising from manual operation and load variations, and the manipulation system minimizes hydraulic control valves of a single function to the utmost and is directed to make unitary member/system functions precise and improve the accuracy and reliability.

Further, since such a distributor 31a as described above is provided, flow rate distribution or control of the hydraulic pump 2 by complicated manual operations of the manually operable levers 30A and 30B upon simultaneous operation which are conventionally operated manually and adjusted by an operator relying upon the experience of the operator itself can be set to desired manners of the operator based on contents of the works. In other words, different priorities can be provided to operations of the actuators 7A and 7B depending upon the contents of the works.

Consequently, the manipulation system can cooperate with the power supply system described above to automatically effect accurate flow rate control irrespective of the loads 8A and 8B only by manually operating the manually operable levers 30A and 30B while the operator places stress on grasping of a load condition of the working machine.

(c) Valve Control System

Subsequently, operation of the valve control system will be described taking notice of the actuator (hydraulic cylinder) 7A with reference to FIG. 3.

First, the actuator flow rate setting signal Qsa outputted from the distributor 31a is inputted to the valve controller 32A. Meanwhile, a flow rate signal Qsaa to the actuator 7A is fed back by a flow rate sensor 106A with a check valve. Then, a signal (P control signal) obtained by multiplying a difference signal between the signal Qsa and the signal Qsaa by a constant Kp, another signal (I control signal) obtained by multiplying an integrated value of the difference signal between the signal Qsa and the signal Qsaa by a constant 1/T and a further signal F(Qsa) which is a feedforward signal of the signal Qsa are added.

It is to be noted that the flow rate of the main control valve 6A may alternatively be calculated from, in place of the flow rate sensor 106A with a check valve, a pressure difference (Ps - P11a or Ps - P12a) across the main control valve

6A, an output Xca of the spool position sensor 107A of the main control valve 6A or the like.

Further, as described hereinabove, the valve control system has a large number of resonance and antiresonance points because the mass loads 8A and 8B which vary over large extents are driven, and particularly since a rocking phenomenon having a low frequency deteriorates the driving feeling, a signal P11a from the A port load pressure sensor 108A of the main control valve 6A and a signal P12a from the B port load pressure sensor 109A of the main control valve 6A are fed back to the valve controller 32A via the band-pass filters 200. In other words, the present system is a dynamic pressure feedback system.

Finally, the main control valve (3-stage amplification type main control valve) 6A can feed back, since the signal Xca of the spool position (spool opening) which increases in proportion to an input current value Xci to the servo valve for the main control valve is obtained from the spool position sensor 107A, this signal Xca to the valve controller 32A to position the spool of the main control valve 6A so that the signal Qsaa which is equal to the actuator flow rate setting signals Qsa can be obtained automatically.

The present system is a servo mechanism of the automatic flow rate control type replacing conventional flow rate adjustment to the actuators 7A and 7B performed by manual operations, and can be improved in terms of the response, safety and accuracy in flow rate.

Subsequently, a modification to the first embodiment of the present invention will be described with reference to FIG. 4. The present modification is constructed in a substantially similar manner to that of the first embodiment described above, and principally, differences thereof from the first embodiment will be described below. The present modification is constructed such that the actuator flow rate setting signals Qsa and Qsb set by the distributor 31a are set for each work mode of the construction machine (for example, an excavation work mode, a house demolition work mode and so forth).

In particular, in the first embodiment described above, when the sum total of requested flow rates to the hydraulic actuators 7A, 7B, ... by the manually operable levers 30A, 30B, ... is higher than the pump delivery flow rate, either ① the first coefficient α is calculated by [pump delivery flow rate]/[sum total of requested flow rates], and the requested flow rates set by the manually operable levers 30A, 30B, ... are multiplied by the first coefficient α to set the actuator flow rate setting signals as $Qsa = \alpha Qra$, $Qsb = \alpha Qrb$, ..., or ② the second coefficient β is calculated by [allowable supply flow rate]/[sum total of requested flow rates], and the requested flow rates set by the manually operable levers 30A, 30B, ... are multiplied by the second coefficient β to set the actuator flow rate setting signals as $Qsa = \beta Qra$, $Qsb = \beta Qrb$, ...

In this instance, the coefficients (first coefficient α or second coefficient β) by which the requested flow rates Qra, Qrb, ... to the actuators 7A and 7B are multiplied have a value equal for all of the actuators 7A and 7B. In particular, in the case of ① described above, all of Qra and Qrb are multiplied uniformly by the first coefficient α, and in the case of ②, all of Qra and Qrb are multiplied uniformly by the second coefficient β.

By the way, since the requested flow rates Qra, Qrb, ... are all set in response to manually operated conditions of the manually operable levers 30A and 30B, while different priorities are applied already to operations of the actuators 7A, 7B, ... depending upon the magnitudes of the set request signals Qra, Qrb, ..., if the first coefficient α or the second coefficient β is set individually for each actuator, then the priorities of the individual actuators can be made definite and the workability is improved. In short, if, depending upon the mode of the work (that is, the work mode), not the requested flow rates Qra, Qrb, ... set in response to manually operated conditions of the manually operable levers 30A and 30B are corrected using the coefficient α (or β) of the same value, but the actuator flow rate setting signals Qsa, Qsb, ... are set using coefficients obtained by weighting the requested flow rates Qra, Qrb, ... in accordance with the mode of the work, then the operability and the workability can be further improved.

Therefore, in the present modification, the coefficient α (or β) by which the requested flow rates Qra and Qrb are multiplied is multiplied by correction coefficients kij which are set for the individual actuators in response to the actuators or the mode of the work (that is, the work mode).

Here, the correction coefficients kij will be described. The correction coefficients kij are set depending upon the actuator (i) and the work mode (j), and can be represented as $kij = F(i, j)$.

In particular, the distributor 31a has such a data table as shown in FIG. 4 set therein, and in the data table, the correction coefficients kij set depending upon the actuator number i and the work mode number j are stored in the form of a table.

Here, the work mode is a work mode set arbitrarily by an operator, and such modes as, for example, an excavation mode for $j = 1$ and a house demolition mode for $j = 2$ are set. The operator can perform setting of a work mode or changing of a set work mode by manually operating a manually operable member in an operator cab, and priorities suitable for each work mode can be set for the individual actuators ($i = 1, 2, ...$).

Meanwhile, i is a number indicating an actuator, and where the construction machine is, for example, a hydraulic shovel, $i = 1$ represents a boom cylinder, $i = 2$ represents a stick cylinder, $i = 3$ represents a bucket cylinder, and $i = 4$ represents a swing motor.

If an operator elects, for example, the excavation mode ($j = 1$) as a work mode, then correction coefficients k_{11}, k_{21}, ... are set for the boom cylinder ($i = 1$), the stick cylinder ($i = 2$), ..., respectively.

For example, if an operator selects the excavation mode (j = 1) as the work mode, then the correction coefficients k_{11}, k_{21}, ... are set for the boom cylinder (i = 1), the stick cylinder (i = 2), ...

Accordingly, when the sum total of the requested flow rates to the hydraulic actuators 7A, 7B, ... by the manually operable levers 30A, 30B, ... is higher than the pump delivery flow rate, ① the first coefficient α is calculated by [pump delivery flow rate]/[sum total of requested flow rates], and the first coefficient α, the requested flow rates Qra, Qrb, ... set by the manually operable levers 30A, 30B, ... and the correction coefficients k_{11}, k_{21}, ... set for the individual actuators are multiplied by each other to set the actuator flow rate setting signals Qsa, Qsb, ... In particular, the actuator flow rate setting signals are set as $Qsa = \alpha \cdot k_{11} \cdot Qra$, $Qsb = \alpha \cdot k_{12} \cdot Qrb$, ...

On the other hand, also ② when the second coefficient β is calculated by [allowable supply flow rate]/[sum total of requested flow rates], the actuator flow rate setting signals are set as $Qsa = \beta \cdot k_{11} \cdot Qra$, $Qsb = \beta \cdot k_{12} \cdot Qrb$, ... in a similar manner as described above.

Then, by setting the actuator flow rate setting signals Qsa and Qsb to be set by the distributor 31a using the correction coefficients kij set individually for each work mode and each actuator of the construction machine, a flow rate distribution to the actuators suitable for the work mode of the construction machine can be realized. Particularly upon simultaneous operations of a plurality of actuators, operation on which a will of an operator is reflected can be realized.

Consequently, operation conforming to the will of the operator can be performed without requiring any skill, and the workability is improved very much.

It is to be noted that, while, in the example described above, the correction coefficients kij by which the first coefficient α and the second coefficient β are multiplied are set to equal values, the correction coefficients kij may otherwise be set different values between the first coefficient α and the second coefficient β.

By the way, in the first embodiment and the modification to it described above, one three way solenoid valve is used for one actuator for each of the main control valves, and the direction and the flow rate of working fluid to be supplied to a hydraulic actuator is controlled by operation control of the three way solenoid valve. However, the present invention is not limited to such apparatus that have the construction just described, and, for example, as shown in FIG. 5, separate control type valve means which employs a plurality of two way solenoid valves 201 to 204 to control supply of working fluid to an actuator 207 and delivery of working fluid from the actuator 207 independently of each other may be provided.

Here, such separate control type valve means as shown in FIG. 5 is provided taking notice of the operation response of the actuator 207 and can perform supply or delivery of working fluid rapidly and accurately by individually controlling the solenoid valves 201 to 204 provided independently of each other.

Meanwhile, reference numeral 205 denotes a velocity sensor, 207 a hydraulic actuator, 208 and 209 denote each a hydraulic pressure sensor, 210 and 211 denote each a valve position sensor, and 212 and 213 denote each a check valve (directional control check valve).

In the present hydraulic circuit, control signals to the solenoid valves 201 to 204 are set by control means not shown based on detection information from the sensors 205, 208 to 211 to control the changeover conditions of the solenoid valves 201 to 204.

It is to be noted that a solenoid valve of the spool type which is superior in response and stability is used for the two way solenoid valves 201 to 204. While a solenoid valve of the poppet valve type having a high liquid tightness may possibly be used for the solenoid valves 201 to 204, it is considered that a solenoid valve of the spool type having a stable response is more suitable.

(2) Description of the Second Embodiment

Subsequently, a second embodiment of the present invention will be described. The present second embodiment is constructed in a similar manner to the first embodiment principally except that the accumulator 5 is omitted as shown in FIG. 6.

Also the unload valve 3, the check valve 4, the supply pressure sensor 104, the accumulator capacity sensor 105 and so forth which are provided incidentally to the accumulator 5 are omitted. Those elements having same reference symbols as those of FIG. 1 applied thereto in FIG. 6 are same elements or substantially same elements as those described in the first embodiment, and detailed description of them is omitted.

Thus, the distributor 31a outputs, when the sum total of requested flow rates of working fluid to the actuators 7A and 7B by manually operated conditions of the manually operable levers 30A and 30B is lower than the delivery flow rate of the hydraulic pump 2, the requested flow rate signals to the actuators 7A and 7B by the manually operable levers 30A and 30B as they are as actuator flow rate setting signals. However, when the sum total of the requested flow rates is higher than the pump delivery flow rate, the requested flow rates to the actuators 7A and 7B are multiplied by a value α ($\alpha < 1$: coefficient) obtained by dividing the pump delivery flow rate by the sum total of the requested flow rates, and results obtained by the multiplication are newly set as working fluid requested amounts. Then, the distributor 31a outputs the requested flow rate signals as actuator flow rate setting signals.

In other words, the distributor 31a which functions as the valve control means outputs actuator flow rate setting sig-

nals Qsa, Qsb, ... to the valve controllers 32A, 32B, ... in response to a situation of the power supply system when actuator flow rate request signals Qra, Qrb, ... from the manually operable levers (manually operable means) 30A and 30B are inputted.

It is to be noted that the actuator flow rate request signals Qra, Qrb, ... are signals set independently of each other, and the priority degrees of working oil to be supplied to the actuators 7A and 7B are set depending upon the magnitudes of the requested flow rates represented by the signals.

Then, when the sum total of the requested flow rates to the hydraulic actuators 7A, 7B, ... by the manually operable levers 30A, 30B, ... is lower than the delivery flow rate of the hydraulic pump 2, the requested flow rate signals to the actuators 7A and 7B by the manually operable levers 30A, 30B, ... are used as they are as actuator flow rate setting signals. In other words, Qsa = Qra , Qsb = Qrb , ...

On the other hand, if the sum total of the requested flow rates is higher than the pump delivery flow rate, the requested flow rates set by the manually operable levers 30A and 30B are multiplied by [pump delivery flow rate]/[sum total of requested flow rates] = α, and values obtained by the multiplication are outputted as actuator flow rate setting signals to the valve controllers 32A, 32B, ... In other words, Qsa = αQra , Qsb = αQrb , ...

Thus, since such a distributor 31a as described above is provided, flow rate distribution or control of the hydraulic pump 2 by complicated manual operations of the manually operable levers 30A and 30B upon simultaneous operation which are conventionally operated manually and adjusted by an operator relying upon the experience of the operator itself can be set to desired manners of the operator based on contents of the works. In other words, different priorities can be provided to operations of the actuators 7A and 7B depending upon the contents of the works.

Consequently, the manipulation system can cooperate with the power supply system described above, and an operator can automatically effect accurate flow rate control irrespective of the loads 8A and 8B only by manually operating the manually operable levers 30A and 30B while placing stress on grasping of a load condition of the working machine.

Subsequently, the valve control system will be described. Also the valve control system is similar to that described hereinabove in connection with the first embodiment.

In particular, description is given taking notice of the actuator (hydraulic cylinder) 7A. First, the actuator flow rate setting signal Qsa outputted from the distributor 31a is inputted to the valve controller 32A. Meanwhile, a flow rate signal Qsaa to the actuator 7A is fed back by a flow rate sensor 106A. Then, a signal (P control signal) obtained by multiplying a difference signal between the signal Qsa and the signal Qsaa by a constant Kp, another signal (I control signal) obtained by multiplying an integrated value of the difference signal between the signal Qsa and the signal Qsaa by a constant 1/T and a further signal F(Qsa) which is a feedforward signal of the signal Qsa are added.

It is to be noted that the flow rate of the main control valve 6A may alternatively be calculated from, in place of the flow rate sensor 106A, a pressure difference (Ps - P11a or Ps - P12a) across the main control valve 6A, an output Xca of the spool position sensor 107A of the main control valve 6A or the like.

Further, also in the apparatus described in connection with the present second embodiment, similarly to that described hereinabove in connection with the first embodiment, the valve control system has a large number of resonance and antiresonance points because the mass loads 8A and 8B which vary over large extents are driven, and particularly since a rocking phenomenon having a low frequency deteriorates the driving feeling, a signal P11a from the A port load pressure sensor 108A of the main control valve 6A and a signal P12a from the B port load pressure sensor 109A of the main control valve 6A are fed back to the valve controller 32A via the band-pass filters 200. In other words, the present system is a dynamic pressure feedback system.

Finally, the main control valve (3-stage amplification type main control valve) 6A can feed back, since the signal Xca of the spool position (spool opening) which increases in proportion to an input current value Xci to the servo valve for the main control valve is obtained from the spool position sensor 107A, this signal Xca to the valve controller 32A to position the spool of the main control valve 6A so that the signal Qsaa which is equal to the actuator flow rate setting signals Qsa can be obtained automatically.

Thus, due to such a construction as described above, flow rate distribution or control of the hydraulic pump 2 by complicated manual operations of the manually operable levers 30A and 30B upon simultaneous operation which are conventionally operated manually and adjusted by an operator relying upon the experience of the operator itself can be set to desired manners of the operator based on contents of the works. In other words, different priorities can be provided to operations of the actuators 7A and 7B depending upon the contents of the works.

Consequently, an operator can automatically effect accurate flow rate control irrespective of the loads 8A and 8B only by manually operating the manually operable levers 30A and 30B while placing stress on grasping of a load condition of the working machine.

Subsequently, a modification to the second embodiment of the present invention will be described. The present modification is constructed in a substantially similar manner to that of the modification to the first embodiment such that, in the second embodiment described above, the actuator flow rate setting signals Qsa and Qsb set by the distributor 31a are set for each work mode of the construction machine (for example, an excavation work mode, a house demoli-

tion work mode and so forth).

In particular, in the second embodiment described above, when the sum total of requested flow rates to the hydraulic actuators 7A, 7B, ... by the manually operable levers 30A, 30B, ... is higher than the pump delivery flow rate, the coefficient α is calculated by [pump delivery flow rate]/[sum total of requested flow rates], and the requested flow rates set by the manually operable levers 30A, 30B, ... are multiplied by the coefficient α to set the actuator flow rate setting signals as $Qsa = \alpha Qra$, $Qsb = \alpha Qrb$, ...

In this instance, the coefficient α by which the requested flow rates Qra, Qrb, ... to the actuators 7A and 7B is multiplied has a value equal for all of the actuators 7A and 7B. In particular, all of Qra and Qrb are multiplied uniformly by the coefficient α.

By the way, since the requested flow rates Qra, Qrb, ... are all set in response to manually operated conditions of the manually operable levers 30A and 30B, while different priorities are applied already to operations of the actuators 7A, 7B, ... depending upon the magnitudes of the set request signals Qra, Qrb, ..., if the coefficient α mentioned above is set individually for each actuator, then the priorities of the actuators can be made definite and the workability is improved. In short, if, depending upon the mode of the work (that is, the work mode), not the requested flow rates Qra, Qrb, ... set in response to manually operated conditions of the manually operable levers 30A and 30B are corrected using the coefficient α of the same value, but the actuator flow rate setting signals Qsa, Qsb, ... are set using coefficients obtained by weighting the requested flow rates Qra, Qrb, ... in accordance with the mode of the work, then the operability and the workability can be further improved.

Therefore, also in the present modification to the second embodiment, the coefficient α by which the requested flow rates Qra and Qrb are multiplied is multiplied by correction coefficients kij which are set for the individual actuators in response to the actuators or the mode of the work (that is, the work mode).

In particular, the distributor 31a also in this instance has such a data table as shown in FIG. 4 set therein, and in the data table, the correction coefficients kij set depending upon the actuator number i and the work mode number j are stored in the form of a table.

The work mode is a work mode set arbitrarily by an operator, and such modes as, for example, an excavation mode for j = 1 and a house demolition mode for j = 2 are set. The operator can perform setting of a work mode or changing of a set work mode by manually operating a manually operable member in an operator cab, and priorities suitable for each work mode can be set for the individual actuators (i = 1, 2, ...).

Meanwhile, i is a number indicating an actuator, and where the construction machine is, for example, a hydraulic shovel, i = 1 represents a boom cylinder, i = 2 represents a stick cylinder, i = 3 represents a bucket cylinder, and i = 4 represents a swing motor.

If an operator selects, for example, the excavation mode (j = 1) as a work mode, then correction coefficients k_{11}, k_{21}, ... are set for the boom cylinder (i = 1), the stick cylinder (i = 2), ..., respectively.

For example, if an operator selects the excavation mode (j = 1) as the work mode, then the correct ion coefficient k_{11}, k_{21}, ... are set for the boom cylinder (i = 1), the stick cylinder (i = 2), ...

Accordingly, when the sum total of the requested flow rates to the hydraulic actuators 7A, 7B, ... by the manually operable levers 30A, 30B, ... is higher than the pump delivery flow rate, the first coefficient α is calculated by [pump delivery flow rate]/[sum total of requested flow rates], and the coefficient α, the requested flow rates Qra, Qrb, ... set by the manually operable levers 30A, 30B, ... and the correction coefficients k_{11}, k_{21}, ... set for the individual actuators are multiplied by each other to set the actuator flow rate setting signals Qsa, Qsb, ... In particular, the actuator flow rate setting signals are set as $Qsa = \alpha \cdot k_{11} \cdot Qra$, $Qsb = \alpha \cdot k_{12} \cdot Qrb$, ...

Then, by setting the actuator flow rate setting signals Qsa and Qsb to be set by the distributor 31a using the correction coefficients kij set individually for each work mode and each actuator of the construction machine, a flow rate distribution to the actuators suitable for the work mode of the construction machine can be realized. Particularly upon simultaneous operations of a plurality of actuators, operation on which a will of an operator is reflected can be realized.

Consequently, operation conforming to the will of the operator can be performed without requiring any skill, and the workability is improved very much.

It is to be noted that, also in the second embodiment and the modification to it, separate control type valve means including the plurality of solenoid valves 201 to 204 as shown, for example, in FIG. 5 may be employed in place of the main control valves 6A and 6B. Then, by such construction, supply of working oil to the actuator 207 and delivery of working oil from the actuator 207 can be controlled independently of each other.

(3) Others

It is to be noted that the present invention is not limited to the embodiments described above, and the present invention can be embodied modifying it in various forms without departing from the scope thereof.

INDUSTRIAL APPLICABILITY OF THE INVENTION

Where the present invention is applied to a construction machine such as a hydraulic excavation machine or a hydraulic shovel, mutual interference between different actuators caused by a variation in pressure can be eliminated and lower harmonics of a construction machine structure can be suppressed, and improvement in operability and augmentation in driving feeling of an operator can be anticipated. Further, by an action of a distributor, an actuator flow rate distribution requested by the operator can be realized accurately irrespective of the loads to the actuators, and improvement in operability, particularly, in simultaneous operability and fine operability, can be anticipated. Accordingly, the plurality of actuators can be driven at the same time in accordance with a will of the operator, and the efficiency in working is improved. Accordingly, the present invention can contribute to improvement in operability or workability of the construction machine, and it is believed that the utility of the invention is very high.

Claims

1. A control apparatus for a construction machine, characterized in that it comprises:

   manually operable means (30A, 30B) manually operable by an operator;
   working fluid supply means including a hydraulic pump (2) driven by a prime mover (1);
   driving means including a plurality of actuators (7A, 7B) driven by working fluid from said working fluid supply means;
   valve means including a plurality of control valves (6A, 6B) interposed between said driving means and said working fluid supply means for controlling said driving means;
   detection means including working fluid supply flow rate detection means (102) for detecting a supply flow rate of the working fluid from said working fluid supply means; and
   valve control means (31) for receiving an operation command from said manually operable means (30A, 30B) and a result of detection from said detection means (102) and controlling said valve means by a distributor function of comparing requested flow rate information (Qra, Qrb) to said actuators (7A, 7B) set by said manually operable means (30A, 30B) with the working fluid supply flow rate information from said working fluid supply means and determining optimal supply flow rates to said actuators (7A, 7B) in response to a result of the comparison.

2. A control apparatus for a construction machine as set forth in claim 1, characterized in that said valve control means (31) includes a distributor (31a) which outputs the requested flow rate signals (Qra, Qrb) to said actuators (7A, 7B) by said manually operable means (30A, 30B) as actuator flow rate setting signals (Qp) when the requested flow rate information (Qra, Qrb) is lower than the working fluid supply flow rate information, but outputs values obtained by multiplying the requested flow rates (Qra, Qrb) to said actuators (7A, 7B) by a coefficient (α) smaller than 1 as actuator flow rate setting signals (Qsa, Qsb) when a sum total of the requested flow rates (Qra, Qrb) is higher than the working fluid supply flow rate (Qp).

3. A control apparatus for a construction machine as set forth in claim 2, characterized in that the coefficient (α) smaller than 1 has information obtained by normalization of the working fluid supply flow rate (Qp) with the sum total of the requested flow rate information (Qra, Qrb).

4. A control apparatus for a construction machine as set forth in claim 2, characterized in that the actuator flow rate setting signals (Qsa, Qsb) set by said distributor (31a) are set for each work mode of said construction machine.

5. A control apparatus for a construction machine as set forth in claim 1, characterized in that said detection means includes manipulation detection means (106A, 106B, 107A, 107B, 108A, 108B, 109A, 109B) for detecting operation conditions of said valve means, and

   said valve control means (31) includes correction means (32A, 32B) for receiving results of the detection from said manipulation detection means (106A to 109B) and correcting the distributor function.

6. A control apparatus for a construction machine as set forth in claim 5, characterized in that said manipulation detection means (106A to 109B) includes spool position sensors (107A, 107B) for measuring and feeding back spool positions of said control valves (6A, 6B), load sensing load pressure sensors (108A, 108B, 109A, 109B) for measuring and feeding back load pressures, and flow rate sensors (160A, 106B) for measuring and feeding back flow rates supplied to said actuators (7A, 7B).

7. A control apparatus for a construction machine as set forth in claim 6, characterized in that each of said load sens-

ing load pressure sensors (108A, 108B, 109A, 109B) includes a band-pass filter (200) at an output portion thereof.

8. A control apparatus for a construction machine as set forth in claim 1, characterized in that said working fluid supply means includes an accumulator (5) for accumulating the working fluid on a delivery side of said hydraulic pump (2).

9. A control apparatus for a construction machine as set forth in claim 8, characterized in that said working fluid supply means includes an unload valve (3) for bypassing a delivery flow rate of said hydraulic pump (2) in a no-load condition when a capacity of said accumulator (5) exceeds a predetermined amount.

10. A control apparatus for a construction machine as set forth in claim 9, characterized in that said unload valve (3) is provided in parallel to a working fluid supply path on the delivery side of said hydraulic pump (2) while said accumulator (5) is provided in parallel at a portion of said working fluid supply path on a downstream side with respect to a connection point of said unload valve (3) to said working fluid supply path, and a check valve (4) for preventing a back flow from said accumulator (5) is interposed in a portion of said working fluid supply path between the connection portion of said unload valve (3) and a connection portion of said accumulator (5) to said working fluid supply path.

11. A control apparatus for a construction machine as set forth in claim 1, characterized in that said manually operable means (30A, 30B) includes a supply pressure setting unit (20) for keeping a pump delivery pressure of said hydraulic pump (2) fixed.

12. A control apparatus for a construction machine as set forth in claim 1, characterized in that said working fluid supply means includes an accumulator (5) for accumulating the working fluid on a delivery side of said hydraulic pump (2), and

said valve control means (31) includes a distributor (31a) which outputs the requested flow rate signals (Qra, Qrb) to said actuators (7A, 7B) by said manually operable means (30A, 30B) as actuator flow rate setting signals (Qsa, Qsa) when a sum total of the requested flow rates (Qra, Qrb) is lower than the working fluid supply flow rate (Qp), but outputs values obtained by multiplying the requested flow rates (Qra, Qrb) to said actuators (7A, 7B) by a first coefficient (α) smaller than 1 as actuator flow rate setting signals (Qsa, Qsb) when the sum total of the requested flow rates (Qra, Qrb) is higher than the working fluid supply flow rate (Qp), and calculates a total of an accumulation supply flow rate of said accumulator (5) and the working fluid supply flow rate (Qp) as an allowable supply flow rate (Qs) and outputs values obtained by multiplying the requested flow rates (Qra, Qrb) to said actuators (7A, 7B) by a second coefficient (β) having information obtained by normalization of the allowable supply flow rate (Qs) with the sum total of the requested flow rates (Qra, Qrb) as actuator flow rate setting signals (Qsa, Qsb).

13. A control apparatus for a construction machine as set forth in claim 12, characterized in that the first coefficient (α) has information obtained by normalization of the working fluid supply flow rate (Qp) with the sum total of the requested flow rates (Qra, Qrb).

14. A control apparatus for a construction machine as set forth in claim 12, characterized in that the actuator flow rate setting signals (Qsa, Qsb) set by said distributor (31a) are set for each work mode of said construction machine.

15. A control apparatus for a construction machine as set forth in claim 1, characterized in that said detection means includes power supply side detection means (100 to 102) for detecting an operation condition of said working fluid supply means, and

said control means includes power supply side control means (26) for receiving a result of the detection from said power supply side detection means (100 to 102) and controlling said working fluid supply means.

16. A control apparatus for a construction machine as set forth in claim 15, characterized in that said power supply side detection means (100 to 102) includes a rotation condition sensor (101) for detecting a rotation condition of said prime mover (1), an output power sensor (100) for detecting an output power condition of said prime mover (1), and a working fluid pressure sensor (102) for detecting a pressure of the working fluid from said working fluid supply means.

17. A control apparatus for a construction machine, characterized in that it comprises:

manually operable means (30A, 30B) manually operable by an operator;
at least one variable delivery liquid pressure pump (2) driven by an engine (1);
a plurality of liquid pressure actuators (7A, 7B) driven by pressure fluid delivered from said variable delivery liq-

uid pressure pump (2);

a plurality of main control valves (6A, 6B) interposed between said liquid pressure actuators (7A, 7B) and said variable delivery liquid pressure pump (2) for controlling flow rates and directions to said liquid pressure actuators (7A, 7B);

an accumulator (5) provided in a liquid path between said variable delivery liquid pressure pump (2) and said main control valves (6A, 6B) for accumulating the pressure fluid;

an unload valve (3) provided in said liquid path between said variable delivery liquid pressure pump (2) and said main control valves (6A, 6B) for allowing bypassing of a delivery flow rate of said variable delivery liquid pressure pump (2) in a no-load condition when a capacity of said accumulator (5) approaches a maximum value of the capacity;

a distributor (31a) including first calculation means for outputting requested flow rate signals to said actuators (7A, 7B) by said manually operable means (30A, 30B) as they are as actuator flow rate setting signals when a sum total of the requested flow rates to said actuators (7A, 7B) by said manually operable means (30A, 30B) is lower than a delivery flow rate of said variable delivery liquid pressure pump (2), but outputting, when the sum total of the requested flow rates is higher than the pump delivery flow rate, values obtained by multiplying the requested flow rates to said liquid pressure actuators (7A, 7B) by a value (α) obtained by dividing the pump delivery flow rate by the sum total of the requested flow rates as actuator flow rate setting signals and second calculation means for multiplying the requested flow rates to said actuators (7A, 7B) by a value (β) obtained by dividing an allowable supply flow rate calculated as a total of an accumulation supply flow rate of said accumulator (5) and the pump delivery flow rate by the sum total of the requested flow rates and outputting results of the multiplication as actuator flow rate setting signals;

a supply pressure setting unit (20) provided for said manually operable means (30A, 30B) for keeping the pump delivery output fixed;

a valve controller (32A) for receiving the actuator flow rate setting signals from said distributor (31a) and supplying operation signals to said main control valves (6A, 6B);

a manipulation side sensor group provided for said valve controller (32A) and including spool position sensors (107A, 107B) for measuring and feeding back spool positions of said main control valves (6A, 6B), load sensing load pressure sensors (108A, 108B, 109A, 109B) with a band-pass filter (200) for measuring and feeding back load pressures, and flow rate sensors (106A, 106B) for measuring and feeding back flow rates supplied to said liquid pressure actuators (7A, 7B);

a power supply side sensor group including an rotation condition sensor (101) for measuring an engine speed, a rack opening sensor (100) for measuring a rack opening of an engine fuel pump, a tilting angle sensor (103) for measuring a pump tilting angle, a delivery pressure sensor (102) for measuring a pump delivery pressure, a supply pressure sensor (104) for measuring a system supply pressure and an accumulator capacity sensor (105) for measuring a capacity of said accumulator (5);

first command means for generating a tilting angle command signal for said variable delivery liquid pressure pump (2) based on a sum of a difference between a pressure set by said supply pressure setting unit (20) and the feedback signal from said supply pressure sensor (104) and an integrated value of the difference;

second command means for selecting a maximum signal from among said supply pressure setting unit (20) and said load sensing load pressure sensors (108A, 108B, 109A, 109B), determining a value obtained by addition of a fixed value to a value of the maximum signal as a command signal when the value of the maximum signal continues for more than a fixed period of time and generating a tilting angle command signal for said variable delivery liquid pressure pump (2) based on a sum of a difference between the command signal and the feedback signal from said supply pressure sensor (104) and an integrated value of the difference;

third command means for generating a signal to open said unload valve (3) to allow bypassing of the delivery flow rate of said variable delivery liquid pressure pump (2) in a no-load condition when a supply pressure rises higher by a certain value than a preset value and the capacity of said accumulator (5) is in the proximity of a maximum value thereof, but to close said unload valve (3) when the supply pressure drops lower by a certain value than the preset value or the capacity of said accumulator (5) drops to a value in the proximity of a minimum value thereof;

fourth command means for generating an allowable tilting angle command signal for said variable delivery liquid pressure pump (2) within a range of an output power of said engine (1) as a function of the output delivery power of said engine (1) and an efficiency characteristic of the engine-pump;

fifth command means for generating a tilting angle command signal for said variable delivery liquid pressure pump (2) for securing a pump flow rate which increases in proportion to a flow rate request of an operator; and

a pump controller (26) for selecting a lowest one of the generated command signals as a tilting angle command signal for said variable delivery liquid pressure pump (2) and positioning a pump tilting angle based on a difference between the selected tilting angle command signal and the feedback signal from said tilting angle sensor.

18. A control apparatus for a construction machine, characterized in that it comprises:

manually operable means (30A, 30B) manually operable by an operator;
a hydraulic pump (2) driven by a prime mover (1);
a plurality of actuators (7A, 7B) driven by working fluid from said hydraulic pump (2);
a plurality of control valves (6A, 6B) for controlling said actuators (7A, 7B); and
valve control means (31) for comparing requested flow rate information to said actuators (7A, 7B) set by said manually operable means (30A, 30B) with working fluid supply flow rate information from said hydraulic pump (2), determining optimal supply flow rates to said actuators (7A, 7B) based on results of the comparison and controlling said valve means with the optimal supply flow rates.

FIG. 1

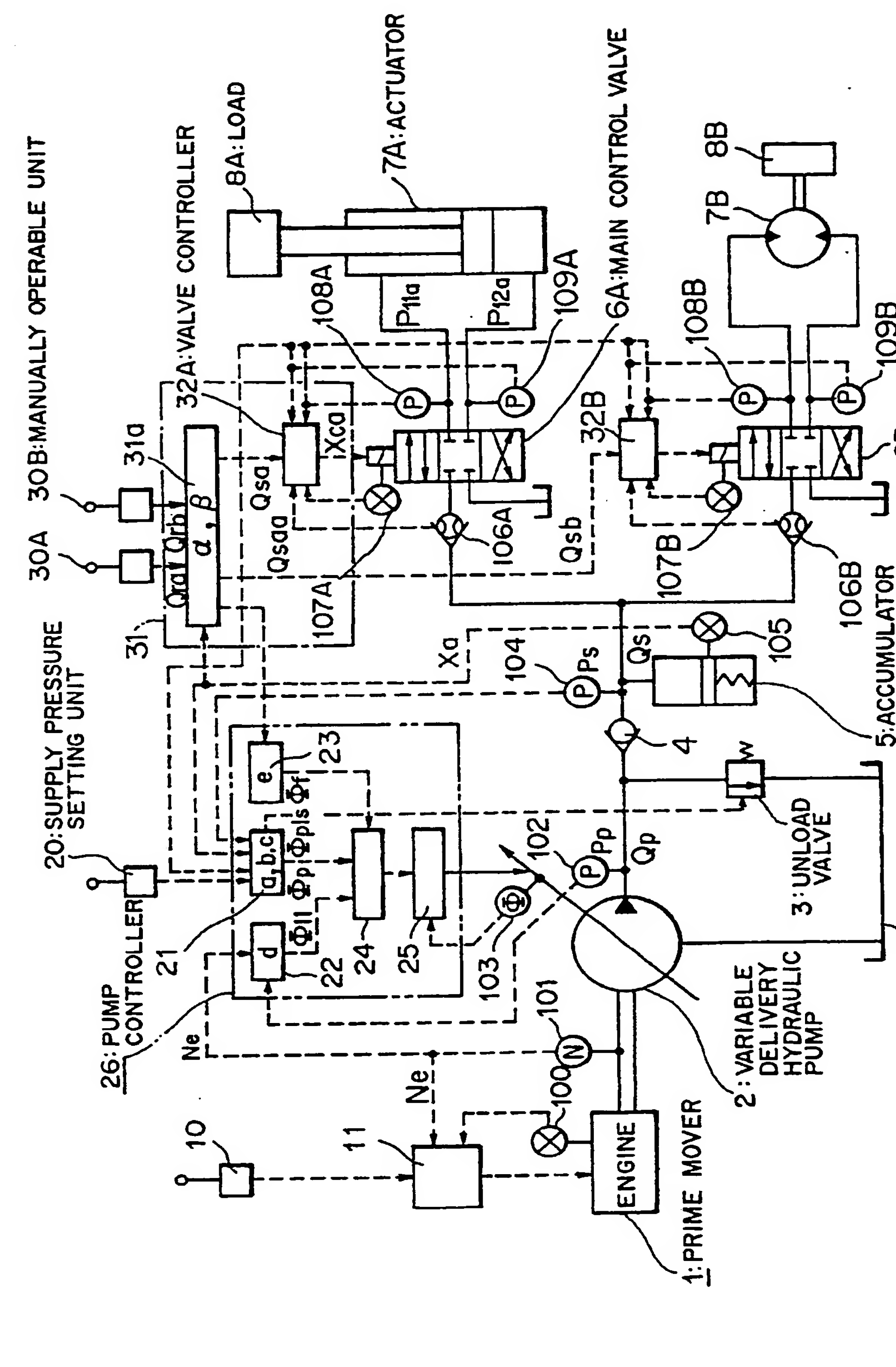

30B:MANUALLY OPERABLE UNIT
32A:VALVE CONTROLLER
8A:LOAD
7A:ACTUATOR
6A:MAIN CONTROL VALVE
108A
109A
108B
109B
32B
8B
7B
6B
30A
31a
31
Qsa
Qsaa
Qsb
Xca
106A
107A
107B
106B
Xa
104
Ps
Qsi
105
5:ACCUMULATOR
20:SUPPLY PRESSURE SETTING UNIT
26:PUMP CONTROLLER
21
22
23
24
25
102
103
Pp
Qp
4
3:UNLOAD VALVE
9
101
100
Ne
10
11
ENGINE
1:PRIME MOVER
2:VARIABLE DELIVERY HYDRAULIC PUMP

FIG. 2

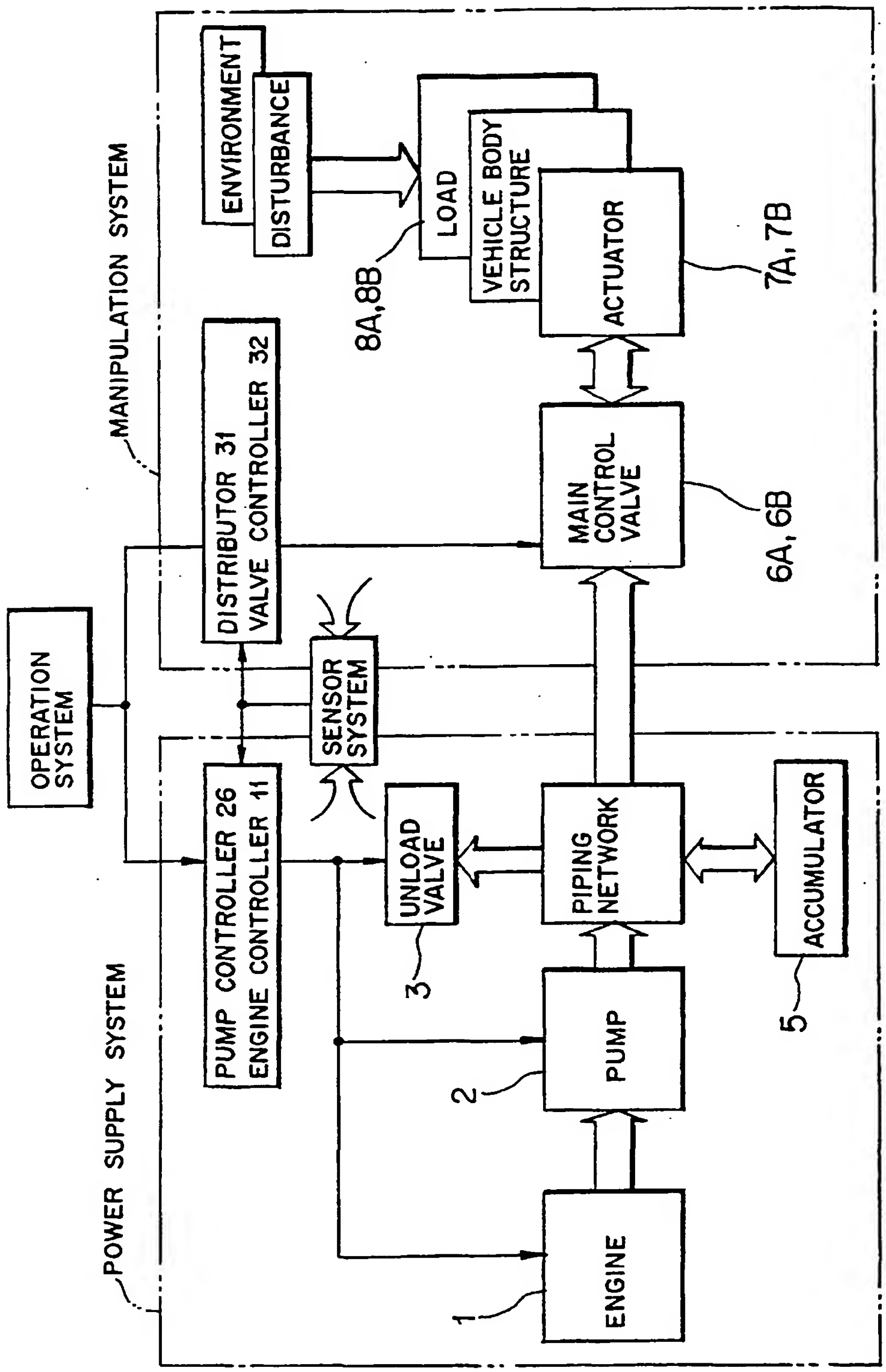
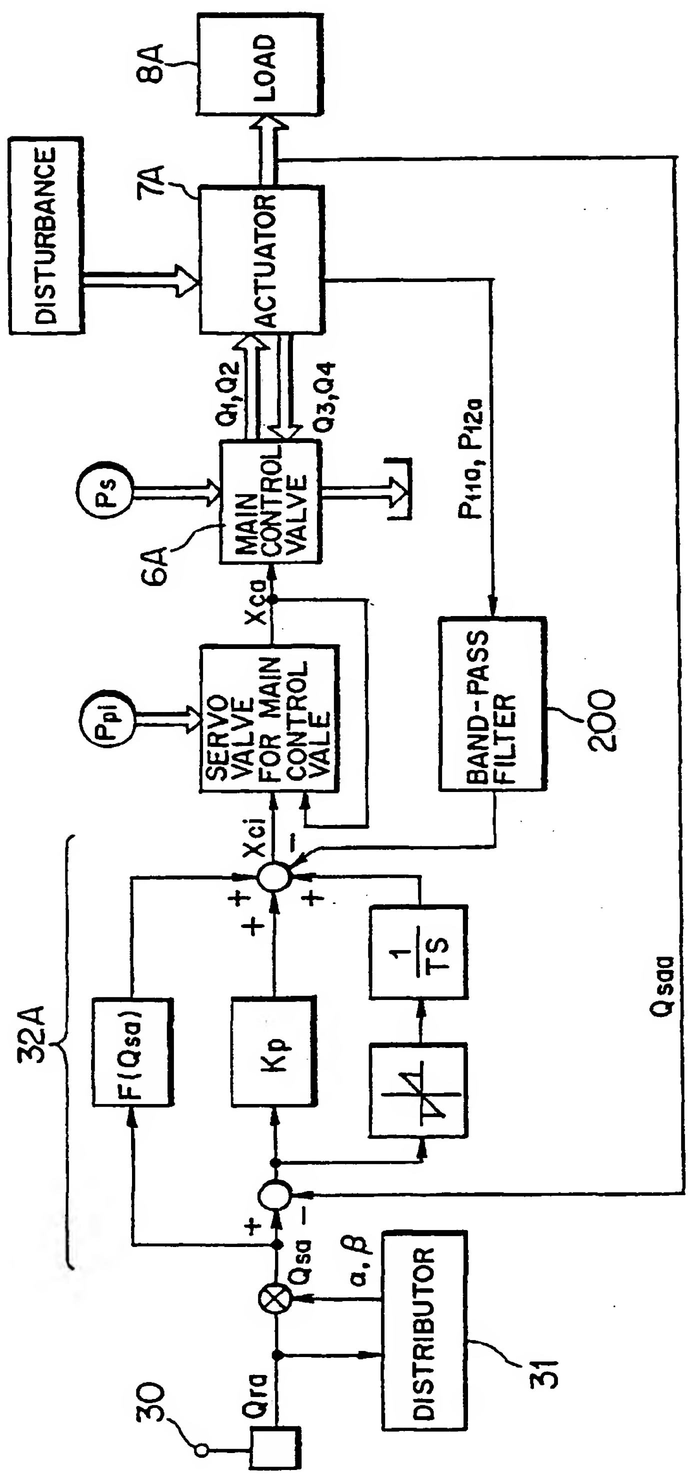

MANIPULATION SYSTEM
POWER SUPPLY SYSTEM
ENVIRONMENT
DISTURBANCE
LOAD
VEHICLE BODY STRUCTURE
ACTUATOR
8A, 8B
7A, 7B
DISTRIBUTOR 31
VALVE CONTROLLER 32
MAIN CONTROL VALVE
6A, 6B
OPERATION SYSTEM
SENSOR SYSTEM
PUMP CONTROLLER 26
ENGINE CONTROLLER 11
UNLOAD VALVE
3
PIPING NETWORK
ACCUMULATOR
5
PUMP
2
ENGINE
1

FIG. 3



DISTURBANCE
LOAD
8A
ACTUATOR
7A
Q_1, Q_2
Q_3, Q_4
P_s
MAIN CONTROL VALVE
6A
X_{ca}
P_{pi}
SERVO VALVE FOR MAIN CONTROL VALE
P_{11a}, P_{12a}
BAND-PASS FILTER
200
X_{ci}
$\frac{1}{TS}$
Q_{saa}
F(Qsa)
Kp
32A
Q_{sa}
a, β
DISTRIBUTOR
31
30
Q_{ra}

FIG. 4

WORK MODE / ACTUATOR	j = 1 (WORK MODE 1)	j = 2 (WORK MODE 2)	j = 3 (WORK MODE 3)
i = 1 : BOOM	K_{11}	K_{12}	K_{13}
i = 2 : STICK	K_{21}	K_{22}	K_{23}
i = 3 : BUCKET	K_{31}	K_{32}	K_{33}
i = 4 : SWING MOTOR	K_{41}	K_{42}	K_{43}

FIG. 5

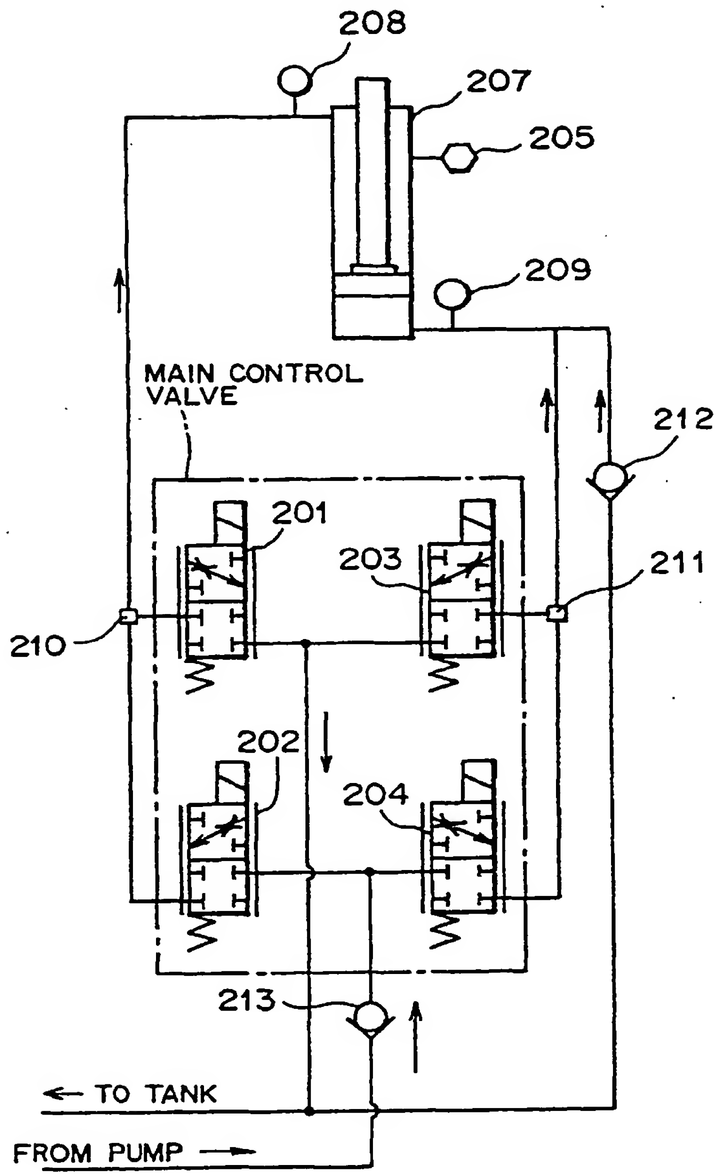

208
207
205
209
MAIN CONTROL
VALVE
212
201
203
211
210
202
204
213
TO TANK
FROM PUMP

FIG. 6

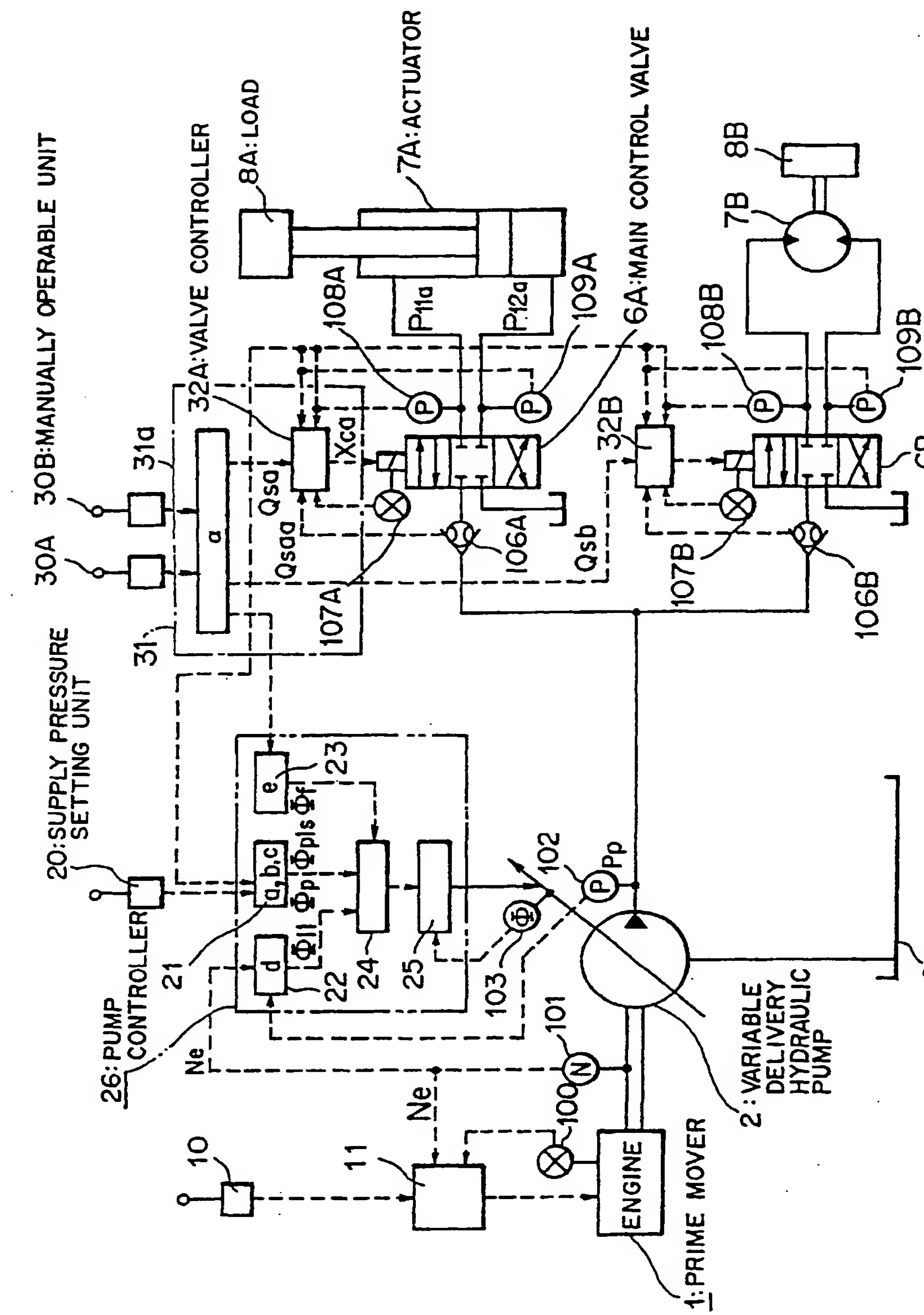

30B:MANUALLY OPERABLE UNIT
32A:VALVE CONTROLLER
8A:LOAD
7A:ACTUATOR
6A:MAIN CONTROL VALVE
30A
31a
31
a
Qsa
Qsaa
Xca
Qsb
32A
32B
108A
109A
108B
109B
106A
107A
106B
107B
P11a
P12a
7B
8B
6B
20:SUPPLY PRESSURE SETTING UNIT
26:PUMP CONTROLLER
a,b,c
d
e
21
22
23
24
25
Φpl
Φpls
Φf
ΦIl
Ne
10
11
100
101
102
103
Pp
9
2:VARIABLE DELIVERY HYDRAULIC PUMP
ENGINE
1:PRIME MOVER

FIG. 7

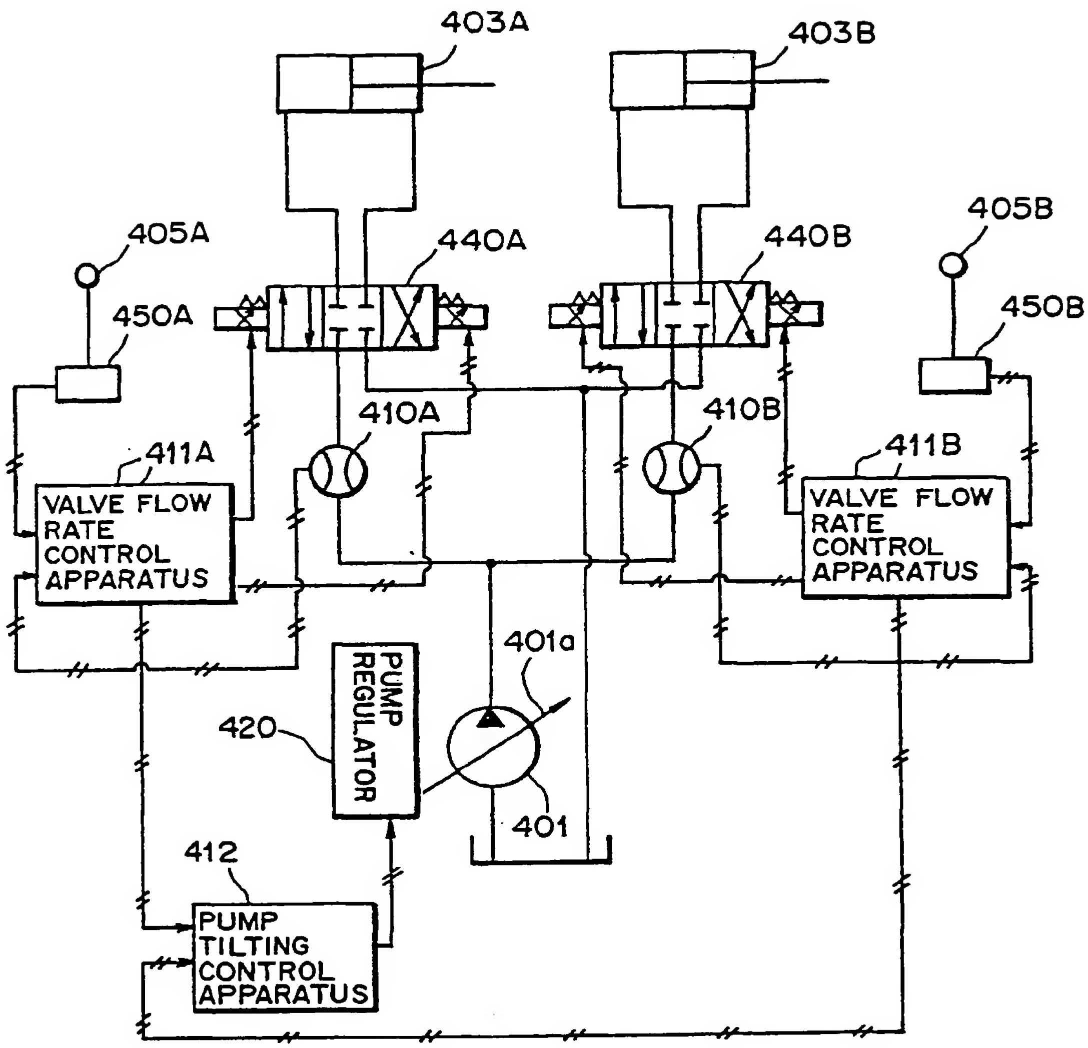
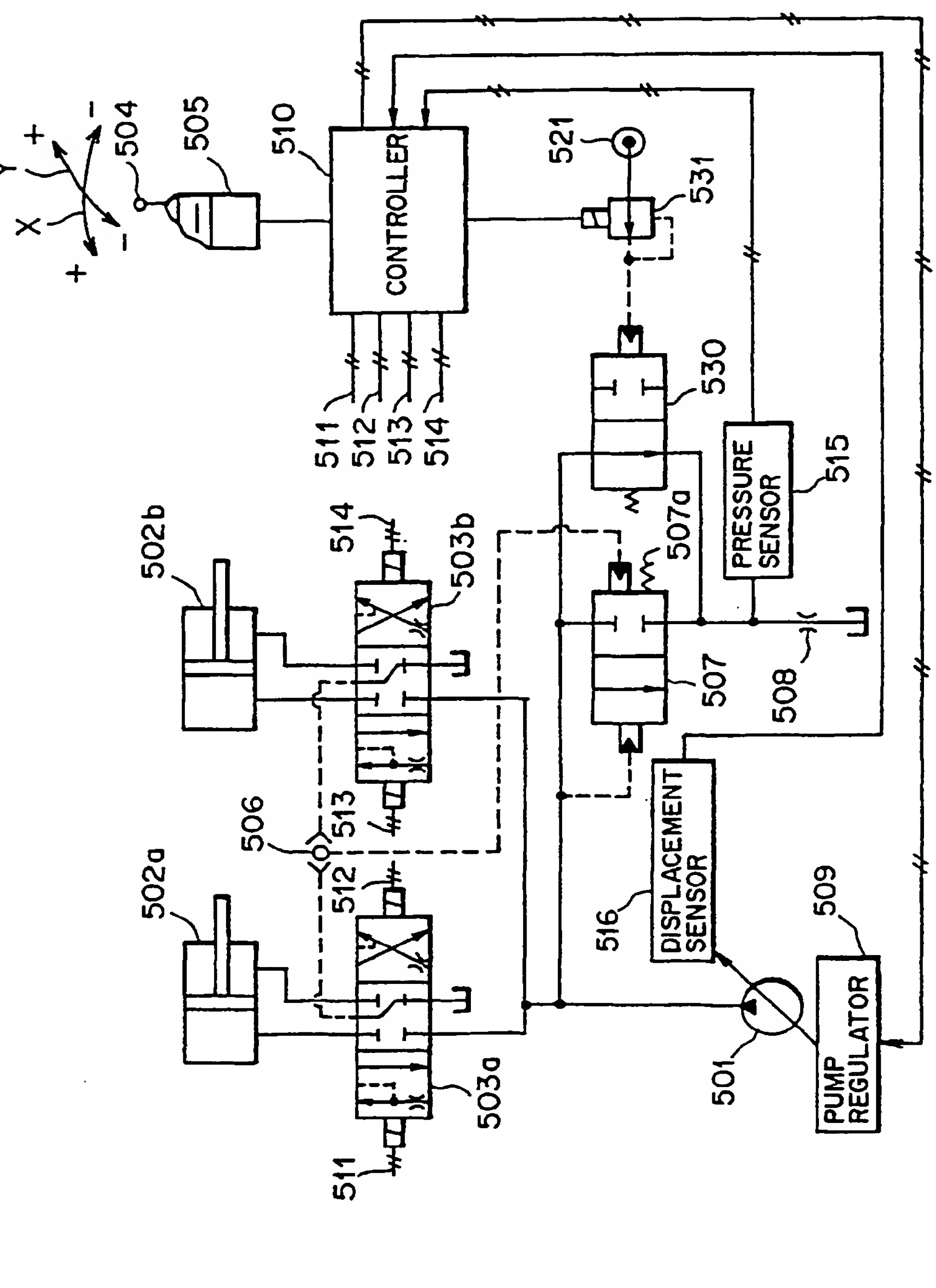

403A
403B
405A
440A
440B
405B
450A
450B
410A
410B
411A
411B
VALVE FLOW RATE CONTROL APPARATUS
VALVE FLOW RATE CONTROL APPARATUS
401a
PUMP REGULATOR
420
401
412
PUMP TILTING CONTROL APPARATUS

FIG. 8


CONTROLLER
PRESSURE SENSOR
DISPLACEMENT SENSOR
PUMP REGULATOR
X
Y
501
502a
502b
503a
503b
504
505
506
507
507a
508
509
510
511
512
513
514
515
516
521
530
531

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP96/02926

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl⁶ E02F9/22, F15B11/00, F15B11/16

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

Int. Cl⁶ E02F9/20-22, F15B11/00-11/12

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Jitsuyo Shinan Koho 1922 - 1996
Kokai Jitsuyo Shinan Koho 1971 - 1996
Toroku Jitsuyo Shinan Koho 1994 - 1996

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	WO, 93/16285, A (Hitachi Construction Machinery Co., Ltd.), August 19, 1993 (19. 08. 93) & US, 5535587, A & EP, 587902, A1	1 - 18
A	WO, 93/18308, A (Hitachi Construction Machinery Co., Ltd.), September 16, 1993 (16. 09. 93) & US, 5394697, A & EP, 597109, A	1 - 18
A	JP, 3-255202, A (Hitachi Construction Machinery Co., Ltd.), November 14, 1991 (14. 11. 91)(Family: none)	1 - 18

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier document but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
December 27, 1996 (27. 12. 96)	January 21, 1997 (21. 01. 97)
Name and mailing address of the ISA/ Japanese Patent Office	Authorized officer
Facsimile No.	Telephone No.

Form PCT/ISA/210 (second sheet) (July 1992)